Apparatus and method for controlling damping force characteristic of vehicular suspension system

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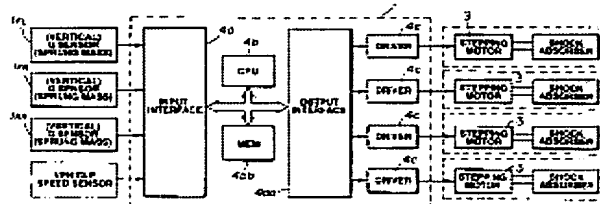- EP0704328 (A3

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Abstract of EP0704327

In apparatus and method for controlling a damping force characteristic for a vehicular suspension system, a relative velocity between a sprung mass and an unsprung mass is calculated using a predetermined transfer function from a sprung mass vertical velocity derived on the basis of a sprung mass acceleration signal. The predetermined transfer function is, in a preferred embodiment, expressed as $GU(S) = (QS^{<2>} + RS)/(FS^{<2>} + DS + E)$, wherein $Q = -m1.c\,2$, $R = -m1.k\,2$, $F = c1.c\,2$, $D = c1.k\,2$, $E = k1.k\,2$, S denotes a Laplace operator, c1 denotes an attenuation constant of each of the shock absorbers, c2 denotes an attenuation constant of the corresponding one of road wheels, k1 denotes a spring constant of the shock absorber, k2 denotes a spring constant of the corresponding one of the road wheels, m1 denotes the sprung mass, and m2 denotes the unsprung mass.

FIG.2

(54) **Apparatus and method for controlling damping force characteristic of vehicular suspension system**

(57) In apparatus and method for controlling a damping force characteristic for a vehicular suspension system, a relative velocity between a sprung mass and an unsprung mass is calculated using a predetermined transfer function from a sprung mass vertical velocity derived on the basis of a sprung mass acceleration signal. The predetermined transfer function is, in a preferred embodiment, expressed as $G_{U(S)} = (QS^2 + RS)/(FS^2 + DS + E)$, wherein $Q = -m_1 \cdot c_2$, $R = -m_1 \cdot k_2$, $F = c_1 \cdot c_2$, $D = c_1 \cdot k_2$, $E = k_1 \cdot k_2$, S denotes a Laplace operator, c_1 denotes an attenuation constant of each of the shock absorbers, c_2 denotes an attenuation constant of the corresponding one of road wheels, k_1 denotes a spring constant of the shock absorber, k_2 denotes a spring constant of the corresponding one of the road wheels, m_1 denotes the sprung mass, and m_2 denotes the unsprung mass.

FIG.2

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Description

BACKGROUND OF THE INVENTION:

Field of The Invention

The present invention relates to apparatus and method for optimally controlling a damping force characteristic of a vehicular suspension system of an automotive vehicle.

Description of Background Art

A Japanese Patent Application First Publication No. Showa 61-163011 exemplifies a first previously proposed suspension system damping force characteristic controlling apparatus, the suspension system having a plurality of shock absorbers.

In the first previously proposed suspension damping force characteristic controlling apparatus, a sprung mass vertical velocity indicative signal and a relative velocity indicative signal between a sprung mass of a vehicle body and unsprung mass of a corresponding road wheel, the sprung mass and unsprung mass being located at a position adjacent to each shock absorber, are detected and outputted.

If direction discriminating signs of the sprung mass vertical velocity indicative signal and the relative velocity indicative signal are coincident with each other, the damping force characteristic of the corresponding one of the shock absorbers is set to a hard characteristic. If either one of the sprung mass vertical velocity signal and the unsprung mass vertical velocity signal is not coincident with the other of both of velocity signals, the damping force characteristic of the corresponding one of the shock absorbers is set to a soft characteristic. In this way, the damping force characteristic controls based on a Sky Hook theorem (theory) are carried out individually for the respective shock absorbers and independently of each other.

In addition, a Japanese Patent Application First Publication No. Heisei 4-63712 exemplifies a second previously proposed suspension system damping force characteristic controlling apparatus, the suspension system having the plurality of shock absorbers.

In the second previously proposed suspension damping force characteristic controlling apparatus, such a shock absorber as having a first mode in which the hard characteristic is exhibited in an extension phase with the soft characteristic exhibited in a compression phase and a second mode in which the hard characteristic is exhibited in the compression phase with the soft characteristic exhibited in the extension phase is used for each shock absorber. Then, a switching control is carried out between the first and second modes according to a direction of the sprung mass vertical velocity Δx (hereinafter, often simply referred to as a sprung mass velocity).

In more details, in the second previously proposed suspension damping force characteristic controlling apparatus, when the direction discriminating sign of the sprung mass vertical velocity signal Δx indicates upward (namely, plus +), the damping force switching control is switched to the first mode so that the hard damping force characteristic is exhibited in the extension phase. When the direction discriminating sign of the sprung mass vertical velocity indicates downward (namely, minus -), the damping force characteristic is switched to the second mode so that the hard damping characteristic is exhibited in the compression phase. Then, the hard damping force characteristic to be controlled in either of the extension or compression phase is further controlled to provide a damping force characteristic position P proportional to the magnitude of the sprung mass velocity Δx so that in a damping suppression region in which both direction discriminating signs of the sprung mass vertical velocity Δx and relative velocity ($\Delta x \dot{-} \Delta x_0$) between the sprung mass and unsprung mass (hereinafter, often simply referred to as a relative velocity) are coincident with each other, the damping characteristic at either one of the extension or compression phase which is required to be controlled at this time of coincidence of direction discriminating signs is further controlled so that its hard damping force characteristic is proportional to the sprung mass velocity Δx and in an oscillation applied region in which both of the direction discriminating signs thereof are not coincident with each other, the damping force characteristic at either one of the extension or compression phase which is required to be controlled is controlled so as to provide a predetermined low (soft) damping force. The second previously proposed suspension damping force controlling apparatus can achieve, with a rather simple construction, such a basic Sky Hook theorem as described above.

In each of the first and second previously proposed suspension damping force controlling apparatuses, the sprung mass vertical velocity Δx is derived by integrating or passing through a low pass filter a sprung mass vertical acceleration signal detected by a sprung mass vertical acceleration sensor installed at the corresponding vehicle body position. In addition, the relative velocity between the sprung mass and unsprung mass is derived by differentiating or passing through a high pass filter a relative displacement indicative signal outputted by a stroke sensor installed on the corresponding unsprung mass.

In the above-described first and second previously proposed suspension system damping force controlling apparatuses, it is necessary to install the stroke sensor onto the unsprung mass portion to detect the relative velocity between the sprung mass and unsprung mass. Therefore, the system configuration of each previously proposed suspension damping force controlling apparatus becomes complicated and it is difficult to actually mount each of the stroke sensors into the vehicle. Furthermore, a cost of manufacturing the suspension system damping force characteristic becomes high.

SUMMARY OF THE INVENTION:

It is an object of the present invention to provide apparatus and method for controlling a damping force of a vehicular suspension system which can achieve a simpler and less expensive construction of the damping force controlling apparatus, a calculation of a relative velocity between a sprung mass of a vehicle body and an unsprung mass of a corresponding one of road wheels (road wheel assemblies) being carried out without each of the relative velocity detecting sensors such as stroke sensors.

According to one aspect of the present invention, there is provided with an apparatus for controlling a damping force characteristic of a vehicular suspension system, said suspension system having shock absorber means interposed between a sprung mass of a vehicle body and unsprung mass of a corresponding one of road wheels, said apparatus comprising: a) an actuator which is so arranged and constructed as to change the damping force characteristic to be exerted by said shock absorber means in response to an input signal; b) detecting means for detecting a behavior of the vehicle body as the sprung mass and outputting a first signal indicating the behavior of the vehicle body; c) velocity converting means for converting the first signal outputted by said detecting means into a second signal indicating a sprung mass vertical velocity of the sprung mass; d) determining means for determining a relative velocity between the sprung mass and unsprung mass and outputting a third signal indicating the determined relative velocity between the sprung mass and unsprung mass; e) control signal generating means for generating and outputting a control signal on the basis of the second signal outputted by said velocity converting means and the third signal outputted by said calculating means; and f) controlling means for controlling the damping force characteristic of said shock absorber means via said actuator on the basis of said control signal, characterized in that said relative velocity determining means comprises calculating means for calculating the relative velocity between the sprung mass (m_1) and the unsprung mass (m_2) using a predetermined transfer function between the first signal outputted by said detecting means and outputting the third signal indicating the calculated relative velocity between the sprung mass and unsprung mass.

According to another aspect of the present invention, there is provided with a method for controlling a damping force characteristic of a vehicular suspension system, said vehicular suspension system having shock absorber means which is interposed between a sprung mass of a vehicle body and an unsprung mass of a corresponding one of road wheels, said method comprising the steps of: a) detecting a behavior of the vehicle body as the sprung mass and outputting a first signal indicating the behavior of the vehicle body; b) converting the first signal outputted by said detecting means into a second signal indicating a sprung mass vertical velocity of the sprung mass; c) calculating a relative velocity between the sprung mass and unsprung mass using a predetermined transfer function from the first signal outputted at said step a) and outputting a third signal indicating the calculated relative velocity between the sprung mass and unsprung mass; and d) generating and outputting the control signal according to the second signal outputted at said step b) and third signal outputted at said step c), said control signal being supplied to damping force characteristic controlling means so that the damping force exerted by said shock absorber means is controlled on the basis of the control signal, characterized in that at said step c), calculating the relative velocity between the sprung mass and unsprung mass using a predetermined transfer function from the first signal outputted at said step a).

BRIEF DESCRIPTION OF THE DRAWINGS:

Fig. 1 is an explanatory view of an arrangement of a vehicular suspension system damping force controlling apparatus in a first preferred embodiment according to the present invention.

Fig. 2 is a circuit block diagram of a control unit and its peripheral circuits of the shock absorber damping force controlling apparatus shown in Fig. 1.

Fig. 3 is a partially sectional view of each shock absorber SA used in the first embodiment shown in Figs. 1 and 2.

Fig. 4 is an enlarged, partially sectional view of the representative shock absorber SA shown in Fig. 3.

Fig. 5 is a characteristic graph representing damping forces with respect to a piston speed of the representative shock absorber SA shown in Figs. 3 and 4.

Fig. 6 is a damping coefficient characteristic graph representing damping force control regions corresponding to stepped positions of an adjuster installed in the representative shock absorber SA associated with a stepping motor according to the rotation of the representative pulse (stepping) motor shown in Figs. 2 and 3.

Figs. 7A, 7B, and 7C are cross sectional views cut away along a line K - K of Fig. 4 representing an essential part of the representative shock absorber shown in Fig. 4.

Figs. 8A, 8B, and 8C are cross sectional views cut away along lines L - L and M - M of Fig. 4 representing an essential part of the representative shock absorber SA shown in Figs. 3 and 4.

Figs. 9A, 9B, and 9C are cross sectional views cut away along a line N - N of Fig. 4 representing an essential part of the representative shock absorber shown in Figs. 3 and 4.

Fig. 10 is a damping force characteristic graph when an extension stroke side (phase) is in a hard damping force characteristic with respect to the piston of the representative shock absorber SA shown in Fig. 4.

Fig. 11 is a damping force characteristic graph when both extension and compression stroke sides (phases) are in soft damping force states.

Fig. 12 is a damping force characteristic graph when the compression stroke side (phase) is in a hard damping force state.

Fig. 13 is a circuit block diagram of a signal processing circuit in the suspension system damping force controlling apparatus in the first embodiment according to the present invention shown in Fig. 1 in which a control signal is formed for each of the shock absorbers.

Fig. 14 is an operational flowchart executed in a control unit in the case of the first embodiment according to the present invention.

Figs. 15A, 15B, 15C, 15D, and 15E are integrally a timing chart indicating a damping force characteristic control operation of a control unit in the first preferred embodiment according to the present invention.

Fig. 16A, 16B, 16C, 16D, and 16E are integrally a timing chart indicating a formed state of the control signal which is formed on the basis of a vertical velocity of a sprung mass and a relative velocity between the sprung mass and an unsprung mass.

Figs. 17A and 17B are graphs representing gain and phase characteristics in the transfer function in the case of the first embodiment, an approximation (approximate) transfer function to the transfer function in an alternative of the embodiment, and an approximation (approximate) band pass filter in an alternative of the embodiment, respectively.

Fig. 18 is an explanatory view of a transfer function calculation model in the damping force controlling apparatus in the case of the first embodiment.

Figs. 19A and 19B are simplified circuit block diagrams of the approximation (approximation) band pass filter in an alternative of the embodiment and the approximation high pass filter in an alternative of the first embodiment of the damping force characteristic controlling apparatus.

BEST MODE CARRYING OUT THE INVENTION:

Reference will hereinafter be made to the drawings in order to facilitate a better understanding of the present invention.

(First Embodiment)

Fig. 1 shows a whole system configuration of a vehicular suspension system damping force characteristic controlling apparatus in a first preferred embodiment according to the present invention.

Four shock absorbers SA_{FL}, SA_{FR}, SA_{RL}, and SA_{RR} (it is noted that subscripts FL denotes a front left road wheel side (position) , FR denotes a front right road wheel side (position), RL denotes a rear left road wheel side (position), RR denotes a rear right road wheel side (position), and a representative shock absorber is simply denoted by SA since all shock absorbers (having the mutually same structures) are interposed between given parts of a vehicular body (sprung mass) and respective road (tire) wheels (unsprung mass). The road wheels

comprise front left road wheel, front right road wheel, rear left road wheel, and rear right road wheels of the vehicle. It is noted that the above-described given parts of the vehicular body indicate front left and right road wheel positions and rear left and right road wheel positions.

As shown in Fig. 1, (vertical, i.e., upward and downward) acceleration (G; gravity) sensors 1 (1_{FL}, 1_{FR}, 1_{RR}) are attached onto given parts of the vehicular body adjacent to the front left and right road wheel side and rear right road wheel side shock absorbers SA , each being provided to detect a vertical sprung mass acceleration acted upon the sprung mass (vehicle body). A control unit 4 is installed at a given part of the vehicle to receive signals derived from the acceleration sensors 1_{FR}, 1_{FL}, and 1_{RR}, processes these signals, and outputs drive signals to respective actuators (,i.e., stepping motors) for the respective four shock absorbers SA.

Fig. 2 shows a circuit block diagram of the vehicular shock absorber damping force controlling apparatus in the first embodiment according to the present invention shown in Fig. 1.

Referring to Figs. 1 and 2, the control unit 4 is installed on a portion of the vehicular body near to a driver's seat. The control unit 4 includes: an input interface circuit 4a; a CPU (Central Processing Unit) 4b; a memory 4bb having a ROM (Read Only Memory) and a RAM (Random Access Memory); an output interface 4aa, and actuator driver circuits 4c; and a common bus.

The control unit 4 is provided with the respective drivers 4c connected between the output interface 4aa and the corresponding stepping motors 3.

The control unit 4 shown in Fig. 2 is provided with a signal processing circuit in terms of its hardware structure as shown in Fig. 13. The explanation of Fig. 13 will be described later.

Next, Fig. 3 show a cross sectional view of each shock absorber SA shown in Figs. 1 and 2.

The shock absorber SA, as shown in Fig. 3, includes: a cylinder 30, a (movable) piston 31 defining an upper portion chamber A and a lower portion chamber B; an outer envelope 33 in which a reservoir chamber 32 is formed on an outer peripheral end of the cylinder 30; a base 34 which defines the lower portion chamber B and the reservoir chamber 32; a guide member 35 which guides a sliding motion of a piston rod 7 with the other end of which the movable piston 31 is linked; a suspension spring 36 interposed between the outer envelope 33 and vehicle body; and a bumper rubber 37.

Each stepping motor 3 shown in Figs. 1 and 2 is installed in an upper position of the corresponding one of the shock absorbers SA, as shown in Fig. 3, so as to operatively rotate an adjuster 40 (refer to Fig. 4) via a control rod 70 in response to a rotation drive signal from the corresponding one of the actuator drivers (circuits) 4c. A rotating shaft of the corresponding one of the stepping motors 3 is mechanically connected to the corresponding adjuster 40 within each shock absorbers SA via the control rod 70.

Fig. 4 shows an enlarged cross sectional view representing a part of the piston assembly 31 and its surrounding part of each of the shock absorbers SA.

As shown in Fig. 4, the piston 31 is formed with penetrating holes 31a and 31b therethrough. In addition, the piston 31 is provided with a compression phase attenuation valve 20 and an extension phase attenuating valve 12, both of the valves 20, 12 respectively opening and closing the respective penetrating holes 31a and 31b. A stud 38 is spirally meshed with and fixed to a bound stopper 41 spirally meshed with and fixed to a tip end of the piston rod 7.

The stud 38 is penetrated through the piston 31. In addition, the stud 38 is formed with a communication hole 39 so as to communicate the upper portion chamber A and the lower portion chamber B, the communication hole 39 forming flow passages (an extension phase second flow passage E, extension phase third flow passage F, bypass flow passage G, and compression phase second flow passage J as will be described later). Then, the adjuster 40 which changes flow passage cross sectional areas of the above-described flow passages is provided within the communication hole 39.

Furthermore, an extension stroke side (phase) check valve 17 and a compression (or contraction) stroke side (compression phase) check valve 22 are also installed on an outer periphery of the stud 38, which enable and disable the fluid flow through the above-described flow passages formed by the communication hole 39 in accordance with a direction of the flow of the fluid. As shown in Fig. 3, the adjuster 40 is rotatable by means of the corresponding one of the actuators (stepping motors) 3 via the control rod 70.

It is noted that the stud 38 is formed with a first port 21, a second port 13, a third port 18, a fourth port 14, and fifth port 16, respectively, in an upper order.

On the other hand, referring to Fig. 4, the adjuster 40 is formed with a hollow portion 19, a first lateral hole 24, and a second lateral hole 25, both lateral holes communicating the internal and external portions of the adjuster 40. A longitudinal groove 23 is formed on an outer peripheral portion. Hence, four flow passages are formed between the upper portion chamber A and lower portion chamber B as the fluid flow passages when the piston stroke indicates the extension phase: that is to say, 1) an extension stroke side (phase) first flow passage D such that the fluid passes the penetrating hole 31b, a valve opened internal side of the extension stroke side (phase) attenuation valve 12, and reaches the lower portion chamber B; 2) an extension stroke side (phase) second flow passage E in which the fluid flows through the second port 13, the longitudinal groove 23, the fourth port 14, a valve opened outer peripheral side of the extension stroke side (phase) attenuation valve 12, and reaches the lower portion chamber B; 3) an extension stroke side (phase) third flow passage F in which the fluid passes through the second port 13, the longitudinal groove 23, and the fifth port 16; and 4) a bypass flow passage G in which the fluid passes through the third

port 18, the second lateral hole 25, and the hollow portion 19 and reaches the lower portion chamber B.

In addition, the three fluid flow passages through which the fluid can be caused to flow during the compression stroke side (phase) of the piston 31 include: 1) a compression stroke side (phase) first flow passage H in which the fluid flows through the penetrating hole 31a and valve opened compression stroke side (phase) attenuation valve 20; 2) a compression stroke side (phase) second flow passage J in which the hollow portion 19, the first lateral hole 24, the first port 21, and the opened compression stroke side (phase) check valve 22 and reaches the upper portion chamber A; and 3) the bypass passage G in which the fluid flows through the hollow portion 19, the second lateral hole 25,, and the third port 18.

In summary, the shock absorber SA is so arranged and constructed as to be enabled to change the damping force characteristics at a multiple stage in its damping characteristic, as shown in Fig. 5, either in the extension phase or compression phase when the adjuster 40 is pivoted according to the rotation of the corresponding one of the stepping motors 3.

Fig. 6 shows relationships between the rotated position of the adjuster 40 and damping force characteristics at both the extension stroke (phase) and compression phase with respect to the piston 31.

In details, as shown in Fig. 6, when the adjuster 40 is pivoted in a given counterclockwise direction from a generally center position at which both of the extension and compression phases are in soft damping force characteristic positions (hereinafter, referred to as a soft region SS), the damping force coefficient at the extension stroke side (phase) can be changed at the multiple stage from a maximum hard to a minimum hard characteristic but the compression stroke side is fixed at a soft position (hereinafter, referred to as an extension stroke side hard region HS). On the contrary, when the adjuster 40 is pivoted in a given clockwise direction therefrom, the damping force coefficient at the compression stroke side (phase) is only changeable to a hard region from the maximum hard to the minimum hard characteristic at the multiple stages and the damping force characteristic at in the compression stroke side is fixed to the soft position (hereinafter, referred to as a compression hard region SH).

When, as shown in Fig. 6, the adjuster 40 is pivoted at any one of positions ①, ②, and ③, cross sections of the piston assembly portions cut away along lines K - K, L - L, M - M, and N - N of Fig. 4 are respectively shown in Figs. 7A (①), 7B (②), and 7C (③) (K-K), 8A (①), 8B (②), and 8C (③) (L - L, M - M), 9A (①), 9B (②), and 9C (③) (N - N), respectively.

The damping force characteristics at the respective positions ①, ②, and ③ shown in Fig. 6 are shown in Figs. 10, 11, and 12, respectively.

Fig. 10 shows the damping force characteristic of the representative shock absorber SA when the adjuster 40 is positioned at ① of Fig. 6.

Fig. 11 shows that when the adjuster 40 is positioned at ② of Fig. 6.

Fig. 12 shows that when the adjuster 40 is positioned at ③ of Fig. 6.

. Next, Fig. 14 shows an operational flowchart for explaining the content of the damping force characteristic control operation for each shock absorber SA executed in the control unit 4.

At a step 101, the CPU 4b determines whether the formed control signal V (for each one of the shock absorbers SA) is increased and exceeds a predetermined positive threshold value δ_T. If Yes at the step 101, the routine goes to a step 102 in which the corresponding one of the shock absorber SA is set to as the extension phase hard region HS.

If NO at the step 101, the routine goes to a step 103 in which the CPU 4b determines whether the control signal V is below a predetermined negative threshold value - δ_C.

If YES at the step 103, the routine goes to a step 104 in which the damping force characteristic of the corresponding one of the shock absorbers SA is set to as the compression phase hard region SH.

If NO at the step 103, the routine goes to a step 105, namely, if the CPU 4b determines that the value of the control signal V falls in a range from $+\delta_T$ to the minus - δ_C, the corresponding one of the shock absorbers SA is set to as each of the respective extension and compression phases being the soft region SS.

Fig. 15A through 15E show integrally a timing chart for explaining the operation of the control unit 4 and shock absorber(s) SA in the case of the first embodiment.

When the control signal V formed on the basis of the sprung mass vertical velocity Δx and relative velocity ($\Delta x - \Delta x_0$) is varied with time as shown in Fig. 15A and the control signal V falls in a range from the predetermined negative threshold value $-\delta_C$ to the predetermined positive threshold value δ_T, the corresponding one of the shock absorbers SA is controlled in the soft region SS.

On the other hand, if the magnitude and direction of the control signal V exceed the predetermined positive threshold value δ_T, the corresponding one of the shock absorbers SA is controlled so that the extension phase hard region HS is provided and the compression phase is set at a predetermined low (soft) damping force characteristic. At this time, the damping force characteristic at the extension phase is increased to provide a target damping force characteristic position P_T in proportion to the magnitude of the control signal V.

On the other hand, if the magnitude and direction of the control signal V are minus and below the predetermined negative threshold value $-\delta_C$, the compression phase hard region SH is provided so that the extension phase damping force characteristic is fixed to the low predetermined damping force characteristic and the damping force characteristic at the compression phase is varied to provide a target damping force characteristic

position P_C in proportion to the value of the control signal V.

Next, a symbol a of Fig. 15C denotes a region in which the direction of the control signal V formed on the basis of the sprung mass vertical velocity Δx and relative velocity ($\Delta x - \Delta x_0$) is inverted from the negative value (downward) to the positive value (upward).

In the region a, the relative velocity ($\Delta x - \Delta x_0$) still provides the negative value (the phase of the shock absorber SA is at the compression phase) so that the corresponding shock absorber SA is controlled at the extension phase hard region HS on the basis of the direction of the control signal V and the phase of the corresponding shock absorber SA is at the extension phase. Kence, at this region a, the extension phase from which the piston 31 of the shock absorber SA is moved away provides the hard characteristic which is proportional to the value of the control signal V.

A region b denotes a region in which the direction (direction discriminating sign) of the control signal V is still positive (upward value) and the relative velocity ($\Delta x - \Delta x_0$) is switched from the negative value to the positive value (extension phase from which the piston of the corresponding shock absorber SA is moved away). At this time, since the shock absorber SA is controlled in the mode of the extension phase hard region HS on the basis of the direction of the control signal V, the stroke direction of the corresponding shock absorber SA is the extension phase. Hence, at the region b, the extension phase side of the shock absorber SA provides the hard characteristic proportional to the value of the control signal V.

A region c denotes a region in which the control signal V is inverted from the positive value (upward) to the negative value (downward). However, at this region c, since the corresponding shock absorber SA is controlled to the compression phase hard region SH on the basis of the direction (direction discriminating sign) of the control signal V, this region c provides the phase (in this region c, the extension phase) at which the piston of the corresponding shock absorber SA is moved with the soft (predetermined low damping force characteristic) characteristic.

A region d denotes a region in which the control signal V is still at the negative value (downward) and the relative velocity ($\Delta x - \Delta x_0$) is changed from the positive value to the negative value (the phase at which the piston of the corresponding shock absorber SA is at the extension phase side). At this time, since the corresponding shock absorber SA is controlled at the compression phase hard region SH on the basis of the direction of the control signal. Hence, the stroke (phase) of the corresponding shock absorber is at the compression phase. In this region d, the (compression) phase at which the piston of the shock absorber SA is moved provides the hard characteristic proportional to the value of the control signal V.

As described above with reference to Figs. 15A through 15C, when the control signal V based on the sprung mass vertical velocity Δx and relative velocity (

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$\Delta x - \Delta x_0$) and the relative velocity of ($\Delta x - \Delta x_0$) have the mutually the same direction discriminating signs (regions b and d), the instantaneous phase at which the piston of the shock absorber SA is moved is controlled at the hard characteristic mode. If the mutual signs thereof (V and ($\Delta x - \Delta x_0$)) are different from each other (regions a and c), the phase at which the piston of the corresponding shock absorber SA is moved is controlled in the soft characteristic. In the first embodiment, the damping force characteristic control based on the Sky Hook theorem is carried out.

In the first embodiment, at a point of time when the phase at which the piston of the corresponding one of the shock absorbers SA is moved is ended, namely, when the region is transferred from the region a to the region b and from the region c to the region d (hard characteristic to the soft characteristic), the damping force characteristic position P_T or P_C at the phase to which the control is switched has already been switched to the hard characteristic side at the previous regions a and c. Consequently, the switching from the soft characteristic to the hard characteristic has been carried out without delay in time.

Next, Fig. 13 shows the configuration of a signal processing circuit for forming the control signal V by which the generation of the control force F based on the Sky Hook theorem (theory) according to the sprung mass vertical velocity Δx and relative velocity ($\Delta x - \Delta x_0$).

At a block B1, the control unit 4 calculates the four road wheel side sprung mass vertical accelerations G_{FL}, G_{FR}, G_{RL}, and G_{RR} on the basis of the vertical acceleration signals of the vehicle body adjacent to the front right and left road wheel side shock absorbers SA_{FL}, SA_{FR}, and the rear right road wheel side shock absorber SA_{RR} detected and outputted from the respective three sprung mass vertical acceleration (G) sensors 1_{FL}, 1_{FR}, and 1_{RR}. G_{RL} is estimated from these signals from the three vertical G sensors.

At a block B2, a vehicular roll acceleration G_R is derived on the basis of the following equation (5) from the front right and left road wheel side vertical sprung mass acceleration signals G_{FL} and G_{FR}.

$$G_R = (G_{FR} - G_{FL})/2 \qquad (1).$$

At a block B3, the control unit 4 integrates or passes through a low pass filter the four road wheel side sprung mass vertical accelerations G_{FL}, G_{FR}, G_{RL}, and G_{RR} and the roll acceleration G_R so as to convert the acceleration signals G_{FL} through G_R to sprung mass vertical velocity x' (V') (V'$_{-FL}$, V'$_{-FR}$, V'$_{-RL}$, V'$_{-RR}$) and the roll velocity V_R, respectively.

At a block B4, the control unit 4 carries out a band pass filter processing in order to increase a signal isolability from those signal components except a target frequency band within which the control of the damping force characteristic is carried out. That is to say, a first band pass filter BPF1 and a second band pass filter BPF2 extract bounce velocity signals V_{BFR}, V_{BFL}, V_{BRR},

and B_{BRL} within a bounce resonance frequency band of the vehicle. In addition, a third band pass filter BPF3 derives a roll velocity signal V_R with a vehicular roll resonance frequency band as a target (a center).

At the next block B5, the bounce coefficients α_f and α_r are set mutually independently of the front road wheel side and rear wheel side bouncing so as to be capable of coping with a difference in a magnitude of vehicular behaviors according to the difference in the specifications (weights, spring constants, and so on) at the vehicular front road wheel side and rear road wheel side.

At a block B6, rolling coefficients r_f and r_r (γ_f and γ_r in Fig. 13) are set independently at the front road wheel side and rear road wheel side so that the control forces against the vehicular behaviors in the rolling direction due to an external force (lateral acceleration) exerted on the vehicle body during a vehicular steering operation do not become insufficient.

At the next block B7, the control unit 4 calculates four sprung mass component signals V_H (V_{FR-RH}, V_{FL-LH}, V_{RR-RH}, V_{RL-LH}):

Front right road wheel side;

$$V_{FR-RH} = \alpha_f \cdot V_{BFR} + r_f \cdot V_R \qquad (2),$$

Front left road wheel side;

$$V_{FR-LH} = \alpha_f \cdot V_{BFL} - r_f \cdot V_R \qquad (3),$$

Rear right road wheel side;

$$V_{RR-RH} = \alpha_f \cdot V_{BRR} + r_r \cdot V_R \qquad (4),$$

Rear left road wheel side;

$$V_{RL-LH} = \alpha_r \cdot V_{BRL} - r_r \cdot V_R \qquad (5).$$

On the other hand, as shown in Fig. 13, the control unit 4 at a block B8 derives the relative velocities between the sprung mass and unsprung mass ($\Delta x - \Delta x_0$) {($\Delta x - \Delta x_0$)$_{FL}$, ($\Delta x - \Delta x_0$)$_{FR}$, ($\Delta x - \Delta x_0$)$_{RL}$, ($\Delta x - \Delta x_0$)$_{RR}$}, respectively, from the four road wheel side sprung mass vertical acceleration signals G_{FL}, G_{FR}, G_{RL}, and G_{RR} calculated at the block B1 using a (predetermined) transfer function $G_{U(S)}$ of the relative velocity ($\Delta x - \Delta x_0$) with respect to the related sprung mass (vertical) acceleration.

In this embodiment, the transfer function is expressed as follows:

$$G_{U(S)} = (QS^2 + RS)/(FS^2 + DS + E) \qquad (6).$$

In the equation (6), S denotes a Laplace operator, $Q = -m_1 \cdot c_2$, $R = -m_1 \cdot k_2$, $F = c_1 \cdot c_2$, $D = c_1 \cdot c_2$, $D = c_1 \cdot k_2 + k_1 \cdot c_2$, and $E = k_1 \cdot k_2$.

Fig. 18 shows an explanatory view for explaining a transfer function calculation model.

As appreciated from Fig. 18, the symbol x_1 denotes a sprung mass input, x_2 denotes an unsprung mass input, x_3 denotes a road surface input from a road sur-

face, m_1 denotes the sprung mass, m_2 denotes the unsprung mass, c_1 denotes an attenuation coefficient of the suspension system, c_2 denotes an attenuation coefficient of the corresponding one of the road wheels, k_1 denotes a spring constant of the suspension system, and k_2 denotes a spring constant of the corresponding one of the road wheels.

At a block B9, the control unit 4 derives the control signal V (extension phase control signals V_{FR-T}, V_{FL-T}, V_{RR-T}, V_{RL-T} and compression phase signals V_{FR-C}, V_{FL-C}, V_{RR-C}, V_{RL-C}) using the following equations (7) through (10) and (7)' through (10)' from the related sprung mass component signal V_H (V_{FR-RH} through V_{RL-LH}) and the relative velocity signal ($\Delta x - \Delta x_0$) formed at the block B1 and B8.

Front right road wheel side;

$$V_{FR-T} = g_f \cdot V_{FR-RH} \cdot (\Delta x - \Delta x_0)_{FR} \quad (7),$$

$$V_{FR-C} = g_f \cdot V_{FR-RH} \cdot (\Delta x - \Delta x_0)_{FR}) \quad (7)',$$

Front left road wheel side;

$$V_{FL-T} = g_f \cdot V_{FR-LH} \cdot (\Delta x - \Delta x_0)_{FL}) \quad (8),$$

$$V_{FL-C} = g_f \cdot V_{FR-LH} \cdot (\Delta x - \Delta x_0)_{FL}) \quad (8)'.$$

Rear right road wheel side;

$$V_{RR-T} = g_r \cdot V_{RR-RH} \cdot (\Delta x - \Delta x_0)_{RR}) \quad (9),$$

$$V_{RR-T} = g_r \cdot V_{RR-RH} \cdot (\Delta x - \Delta x_0)_{RR}) \quad (9)',$$

Rear left road wheel side;

$$V_{RL-T} = g_r \cdot V_{RL-LH} \cdot (\Delta x - \Delta x_0)_{RL}) \quad (10),$$

$$V_{RL-C} = g_r \cdot V_{RL-LH} \cdot (\Delta x - \Delta x_0)_{RL} \quad (10)',$$

wherein g_f denotes a total gain of control for the front road wheel side damping force and g_r denotes a total gain of control for the rear road wheel side damping force.

It is noted that although, in Fig. 13, each relative velocity ($\Delta x - \Delta x_0$) is directly supplied to the block B9, a smoothing circuit enclosed by a phantom line may alternatively be interposed between the blocks B8 and B9.

Next, Fig. 16A through 16E show integrally a timing chart for explaining the formed state of the control signal V in the exemplified signal processing circuit shown in Fig. 13.

In this case, the sprung mass vertical velocity Δx is varied with time at a relatively low frequency form as shown in Fig. 16A and the relative velocity ($\Delta x - \Delta x_0$) at the block B8 in Fig. 13 is varied with time in a relatively high frequency form as shown in Fig. 16B.

As shown in Fig. 16C, the control signal V is formed on the basis of the following equation ($V = (\Delta x)/(\Delta x - \Delta x_0)$, resulting the control signal form

shown in Fig. 16C, so that an upper limit and a lower limit is placed at the plus side and minus side of the control signal V. Thus, the signal form of the control signal approaches to that of Fig. 15A.

In Fig. 16D, in the damping force characteristic position P, a first mode corresponds to, so called, extension phase hard characteristic and compression phase soft characteristic (HS control mode in Fig. 14) and a second mode corresponds to, so called, extension phase soft characteristic and compression phase hard characteristic (SH control mode in Fig. 14). Consequently, the damping force (C) as shown in Fig. 16E is exhibited.

As described above, the suspension system damping force controlling apparatus in the first embodiment has the following advantages:

(1) Since relative velocity detecting sensors such as the stroke sensors, installed on the unsprung mass portion for detecting the relative velocity between the sprung mass and unsprung mass can be omitted, the system configuration of the damping force controlling apparatus can be simplified, an easiness in mounting the apparatus in the vehicle can be improved, and a reduction of manufacturing the damping force controlling apparatus can be achieved.

(2) It is possible to generate the control force in each shock absorber based on the Sky Hook theorem.

(Alternatives)

The present invention does not limit to the above-described embodiment and the various modifications may be made without departing from the scope of the appended claims.

For example, although, in the above-described embodiments, the three sprung mass G sensors (1_{FL}, 1_{FR}, and 1_{RR}) are installed at vehicle body of the front left and right road wheel sides and rear right road wheel sides, the number of installed positions is arbitrary. The suspension system damping force controlling apparatus according to the present invention can be applied to that system in which the single sprung mass (vertical) G (G means gravity) sensor is only installed at the front road wheel side. In this case, the other sprung mass accelerations are calculated on the basis of the output signal of the single G sensor.

Furthermore, in the first embodiment, the above-described transfer function equation (6) is used as the transfer function to derive the relative velocity ($\Delta x - \Delta x_0$) from the related sprung mass acceleration signal. However, the following lower-order (approximation) transfer function G_S (refer to equation (11)) may alternatively be used since the capacity of the programming becomes large and the discrete equation becomes complex when the above-described higher-order transfer function equation (10) is used.

$$G_S = RS/(DS + E) \quad (11).$$

Alternatively, another approximation function or an approximation filter such as a normally used band pass filter BPF or a high pass filter HPF may be used as shown in Fig. 19A or 19B. In Fig. 19A, the frequency band to pass the related sprung mass vertical acceleration signal through this alternative band pass filter is from 9Hz to 13Hz. In Fig. 19B, the frequency band to pass the related sprung mass vertical acceleration signal through this alternative high pass filter is 10Hz or higher.

These approximation function and filter are such that gain and phase characteristics in the frequency band requiring the damping force characteristic control are not largely varied. For the gain and phase characteristics of the approximate filter, refer to Figs. 17A and 17B.

It is noted that the high pass filter HPF may be used since its general equation shown in the following equation (12) is similar to the approximation transfer function shown in the above-described equation (11).

$$\text{HPF (first order) equation} = S/(S+\omega) \quad (12),$$

wherein $\omega = 2\pi f_c$ (f_c: cut-off frequency).

Finally, the front and rear road wheel side control gains g_f and g_r in the above-described equations (7) through (10) and (7)' through (10)' to derive the control signal V may be varied according to the vehicle speed detected by a vehicle speed sensor (denoted by a phantom line) shown in Fig. 2.

Claims

1. An apparatus for controlling a damping force characteristic of a vehicular suspension system, said suspension system having shock absorber means interposed between a sprung mass of a vehicle body and unsprung mass of a corresponding one of road wheels, said apparatus comprising:

   a) an actuator which is so arranged and constructed as to change the damping force characteristic to be exerted by said shock absorber means in response to an input signal;
   b) detecting means for detecting a behavior of the vehicle body as the sprung mass and outputting a first signal indicating the behavior of the vehicle body;
   c) velocity converting means for converting the first signal outputted by said detecting means into a second signal indicating a sprung mass vertical velocity of the sprung mass;
   d) determining means for determining a relative velocity between the sprung mass and unsprung mass and outputting a third signal indicating the determined relative velocity between the sprung mass and unsprung mass;
   e) control signal generating means for generating and outputting a control signal on the basis of the second signal outputted by said velocity

converting means and the third signal outputted by said calculating means; and
   f) controlling means for controlling the damping force characteristic of said shock absorber means via said actuator on the basis of said control signal,
      characterized in that
      said relative velocity determining means comprises calculating means for calculating the relative velocity between the sprung mass (m_1) and the unsprung mass (m_2) using a predetermined transfer function between the first signal outputted by said detecting means and outputting the third signal indicating the calculated relative velocity between the sprung mass and unsprung mass.

2. An apparatus for controlling a damping force characteristic of a vehicular suspension system as claimed in claim 1, characterized in that said predetermined transfer function is expressed as $G_{U(S)} = (QS^2 + RS)/(FS^2 + DS + E)$, wherein $Q = -m_1 \cdot c_2$, $R = -m_1 \cdot k_2$, $F = c_1 \cdot c_2$, $D = c_1 \cdot k_2$, $E = k_1 \cdot k_2$. S denotes a Laplace operator, c_1 denotes an attenuation constant of the shock absorber means, c_2 denotes an attenuation constant of the corresponding one of the road wheels, k_1 denotes a spring constant of the shock absorber means, k_2 denotes a spring constant of the corresponding one of the road wheels, m_1 denotes the sprung mass, and m_2 denotes the unsprung mass.

3. An apparatus for controlling a damping force characteristic of a vehicular suspension system as claimed in claim 1, characterized in that said predetermined transfer function used in said calculating means is expressed as follows: $G_S = RS/(DS+ E)$, wherein S denotes a Laplace operator, $R = -m_1 \cdot k_2$, m_1 denotes the sprung mass, k_2 denotes a spring constant of the corresponding one of vehicular road wheels, $D = c_1 \cdot k_2 + k_1 \cdot c_2$, c_1 denotes an attenuation constant of the shock absorber means, c_2 denotes an attenuation constant of the corresponding one of the road wheels and $E = k_1 \cdot k_2$.

4. An apparatus for controlling a damping force characteristic of a vehicular suspension system as claimed in claim 1, characterized in that said calculating means comprises a filter having gain and phase characteristics approximate to said predetermined transfer function.

5. An apparatus for controlling a damping force characteristic of a vehicular suspension system as claimed in claim 4, characterized in that said filter is a high pass filter having the transfer function as HPF first order equation = $S/(S +\omega)$, wherein

$\omega = 2\pi f_c$ (f_c denotes a cut-off frequency of said high pass filter).

6. An apparatus for controlling a damping force characteristic of a vehicular suspension system as claimed in claim 4, characterized in that said filter is a band pass filter (BPF) passing the first signal in a frequency band requiring the damping force characteristic control by said controlling means.

                             5

                             10

7. A method for controlling a damping force characteristic of a vehicular suspension system, said vehicular suspension system having shock absorber means which is interposed between a sprung mass of a vehicle body and an unsprung mass of a corresponding one of road wheels, said method comprising the steps of:

                             15

    a) detecting a behavior of the vehicle body as the sprung mass and outputting a first signal indicating the behavior of the vehicle body;

                             20

    b) converting the first signal outputted by said detecting means into a second signal indicating a sprung mass vertical velocity of the sprung mass;

                             25

    c) calculating a relative velocity between the sprung mass and unsprung mass using a predetermined transfer function from the first signal outputted at said step a) and outputting a third signal indicating the calculated relative velocity between the sprung mass and unsprung mass; and

                             30

    d) generating and outputting the control signal according to the second signal outputted at said step b) and third signal outputted at said step c), said control signal being supplied to damping force characteristic controlling means so that the damping force exerted by said shock absorber means is controlled on the basis of the control signal,

                             35

                             40

       characterized in that

       at said step c), calculating the relative velocity between the sprung mass and unsprung mass using a predetermined transfer function from the first signal outputted at said step a).

                             45

                             50

                             55

FIG.1

FIG.2

FIG.3

_SA_
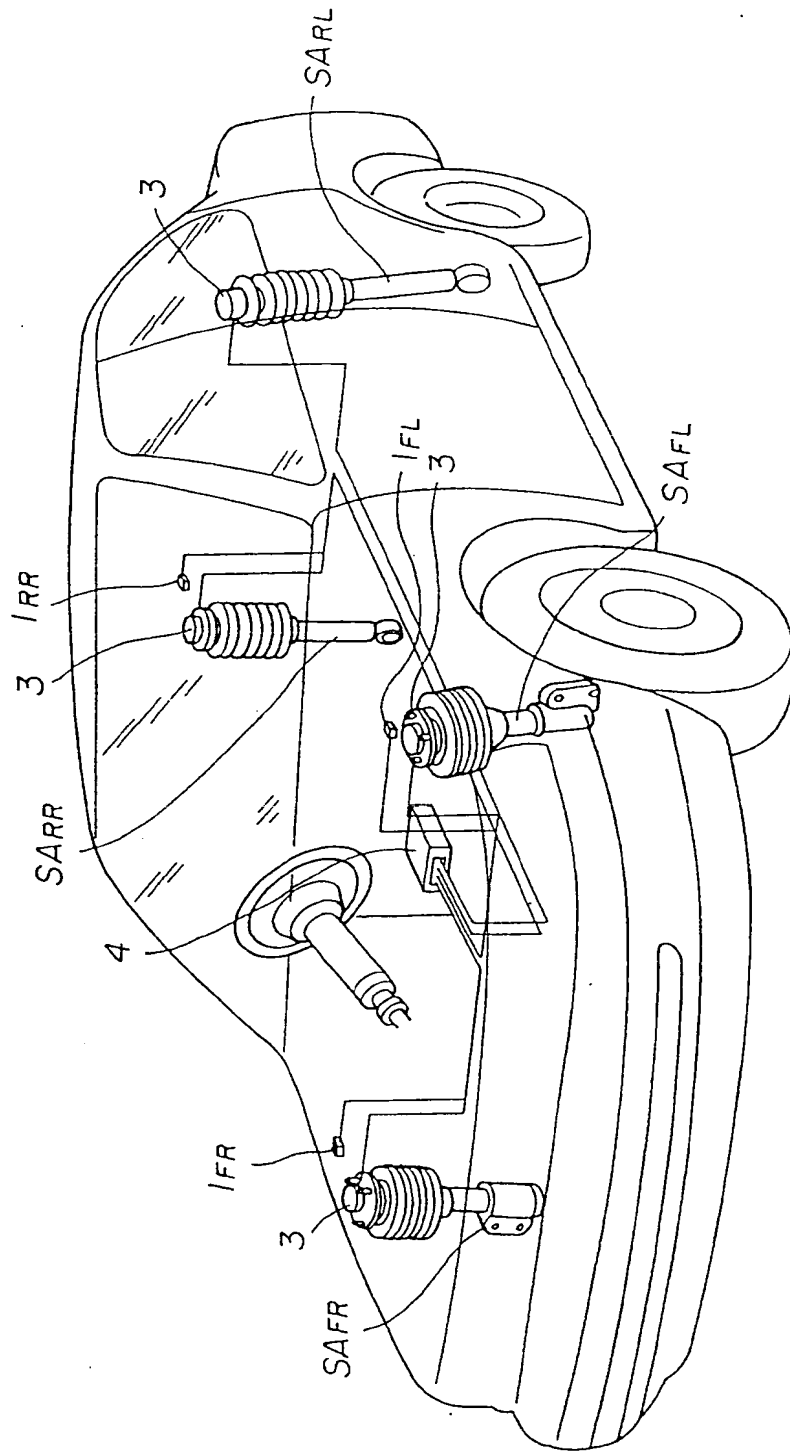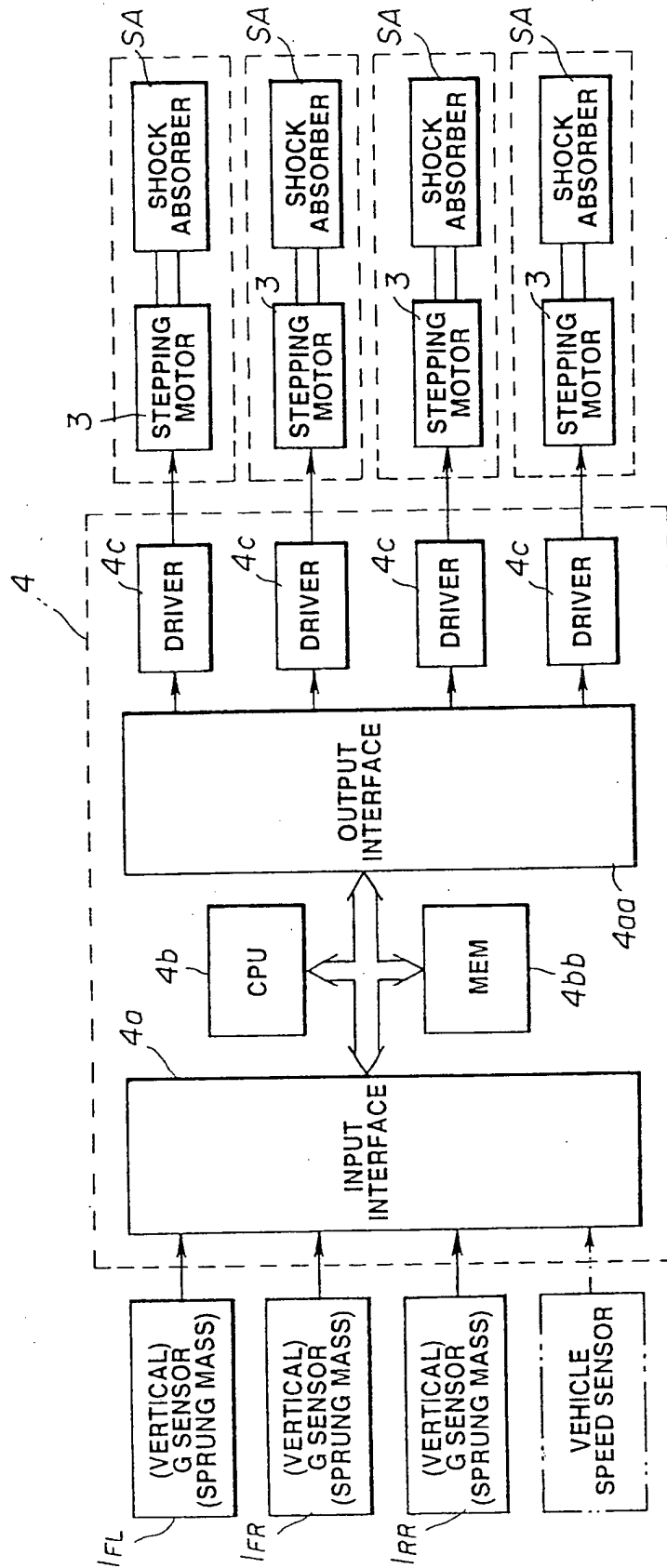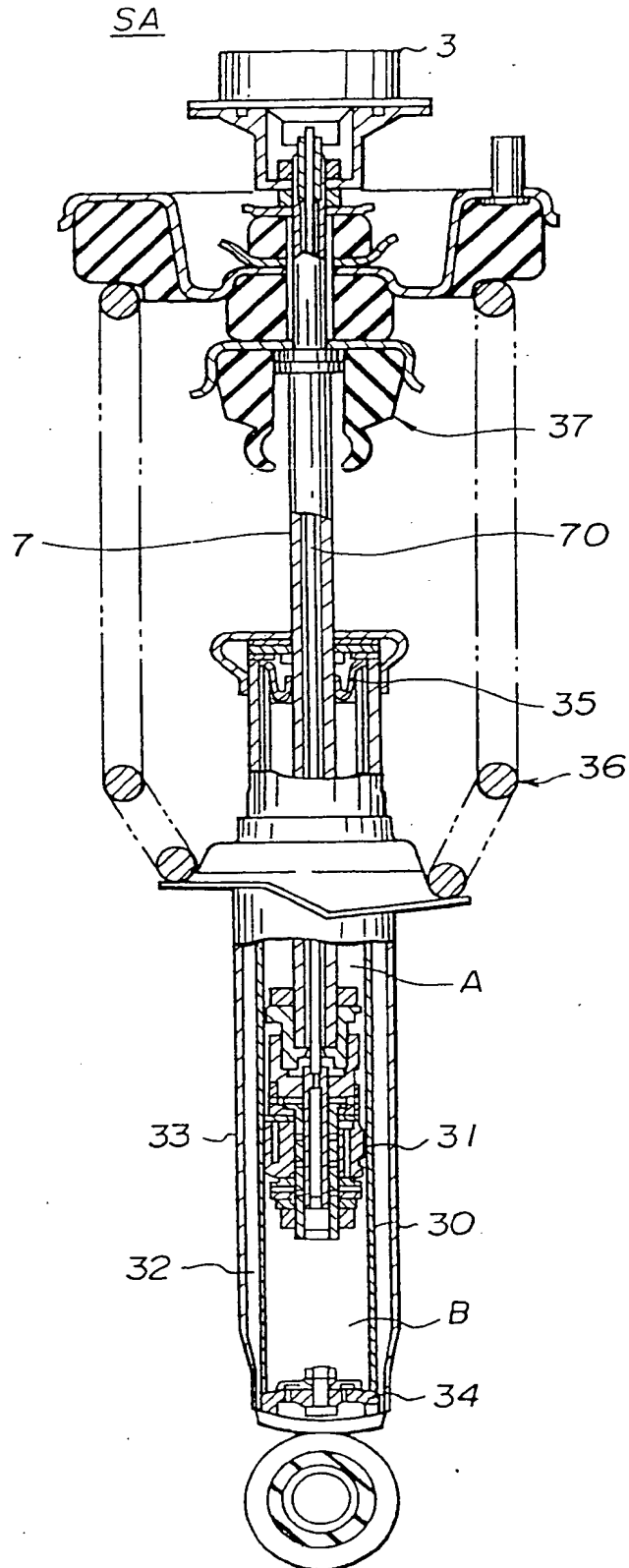
FIG.4
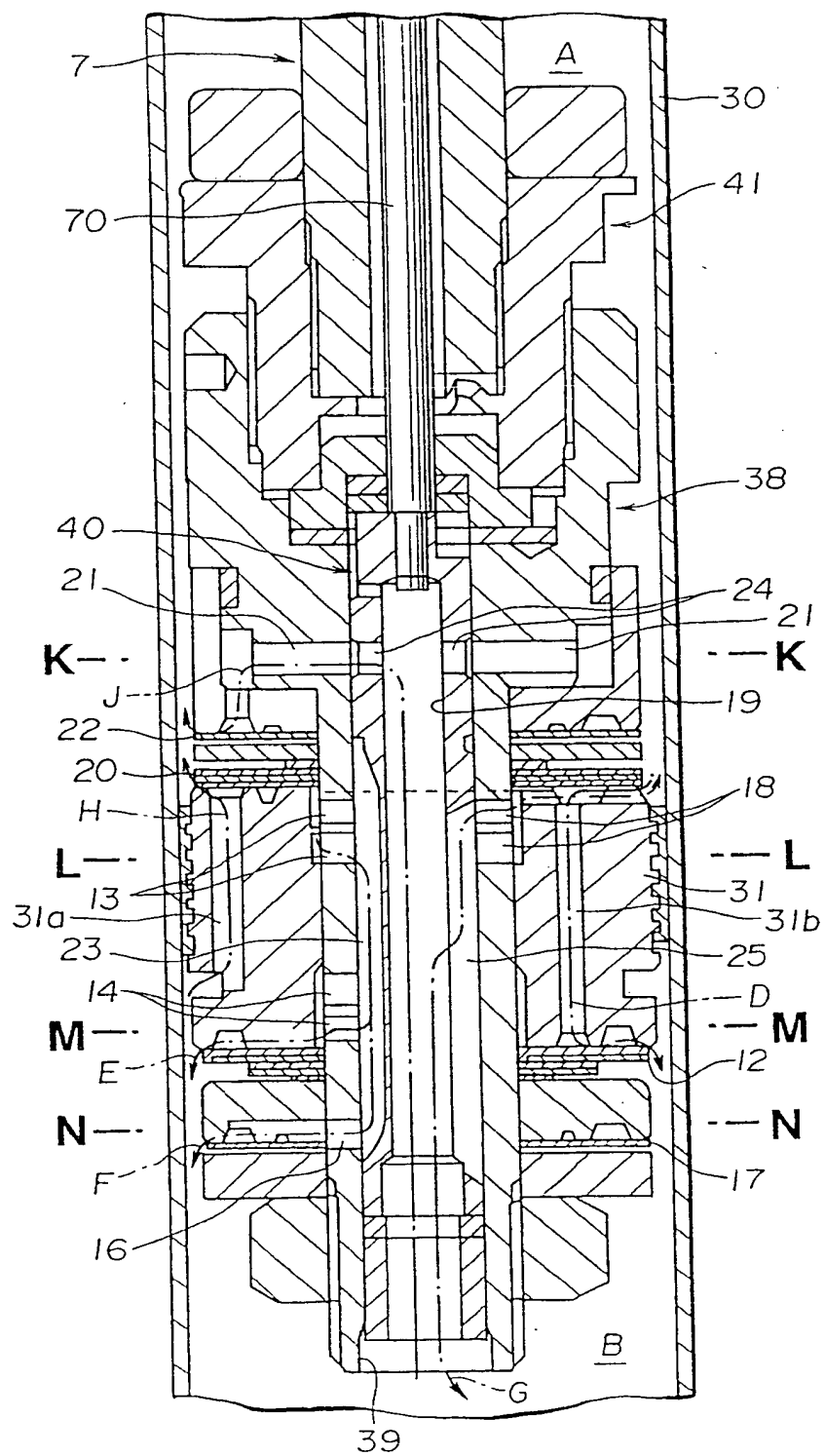
FIG.5
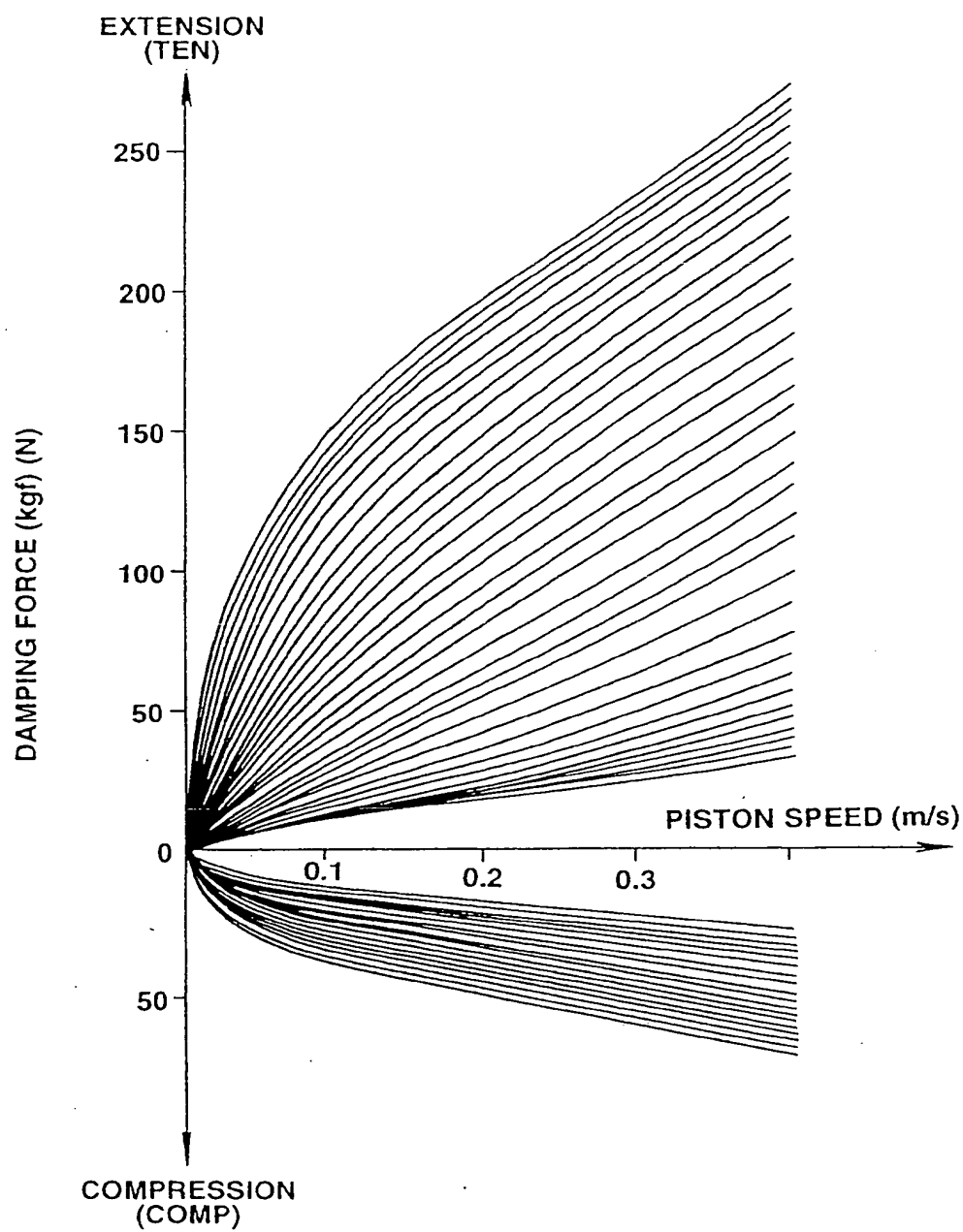
FIG.6
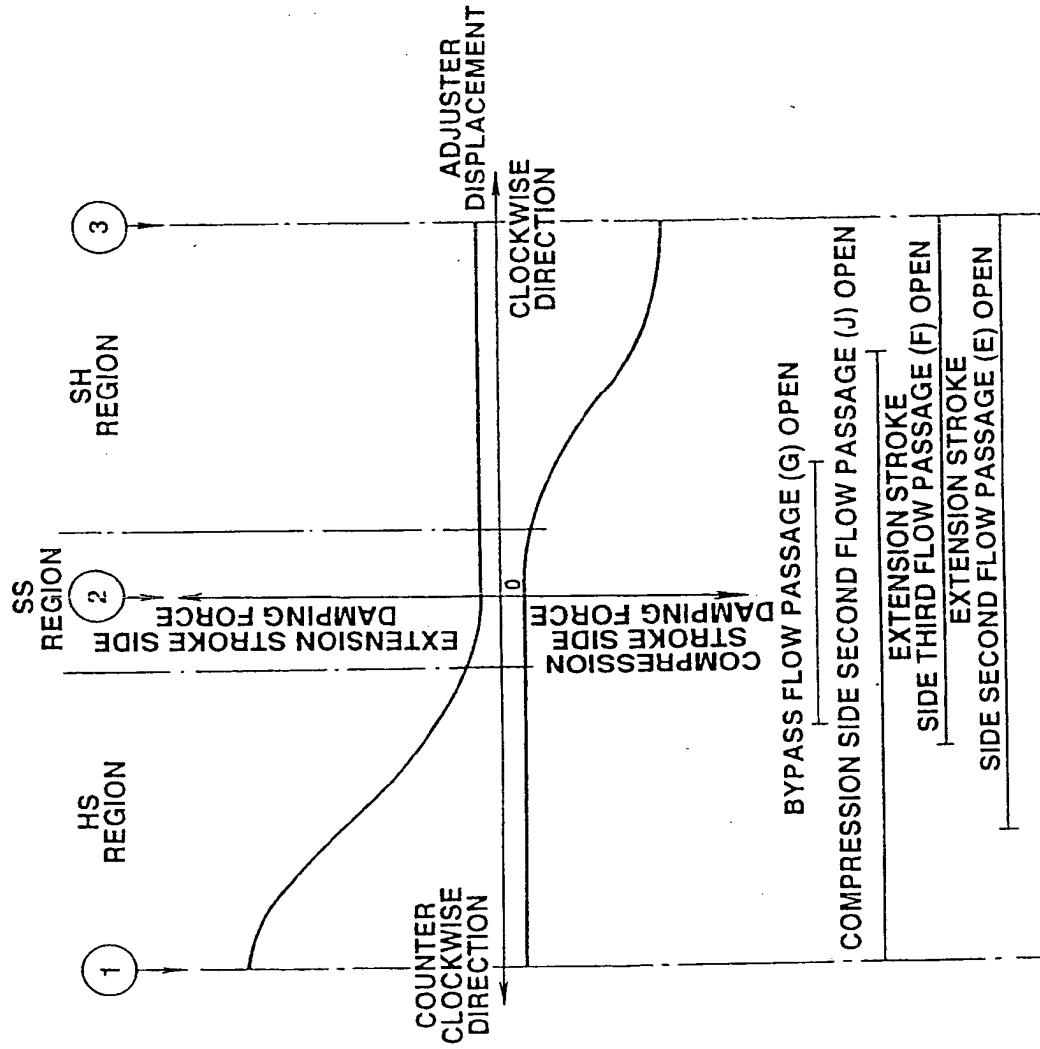
FIG.7A
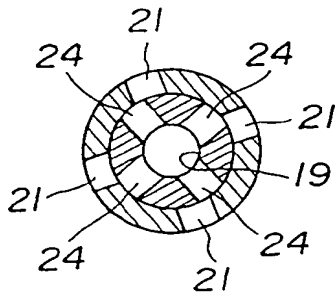
FIG.7B
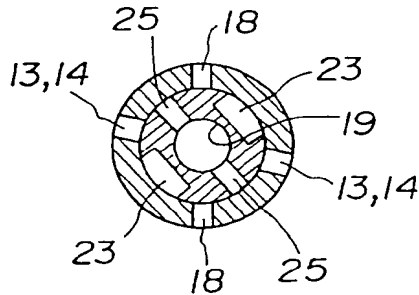
FIG.7C
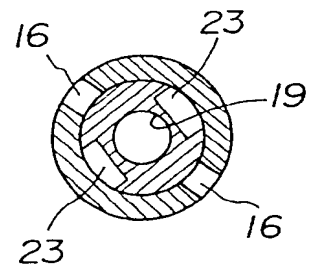
FIG.8A
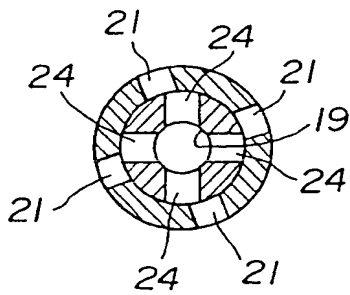
FIG.8B
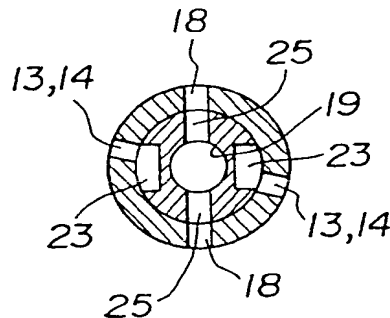
FIG.8C

FIG.9A

FIG.9B

FIG.9C

FIG.10

EXTENSION
(ten)

0

PISTON SPEED

COMPRESSION
(comp)

FIG.11

EXTENSION
(ten)

0

PISTON
SPEED

COMPRESSION
(comp)

FIG.12

EXTENSION
(ten)

0

PISTON SPEED

COMPRESSION
(comp)

FIG.13

FIG.14


```
                START

                 │
                 ▼
            ╱────────╲      101
           ╱          ╲
          ╱  V>δT ?    ╲ ──────── NO ──────────┐
           ╲          ╱                        │
            ╲────────╱                         │
                 │                             ▼
                YES                       ╱────────╲      103
                 │                       ╱          ╲
                 │                      ╱  V<-δc ?   ╲ ──── NO ──────────┐
                 │                       ╲          ╱                    │
                 │                        ╲────────╱                     │
                 │                             │                         │
         102     │                            YES      104              105
                 │                             │                         │
                 ▼                             ▼                         ▼
        ┌──────────────┐            ┌──────────────┐          ┌──────────────┐
        │ HS CON. MODE │            │ SH CON. MODE │          │ SS CON. MODE │
        └──────────────┘            └──────────────┘          └──────────────┘
                 │                             │                         │
                 │◄────────────────────────────────────────────────────┘
                 ▼
               END
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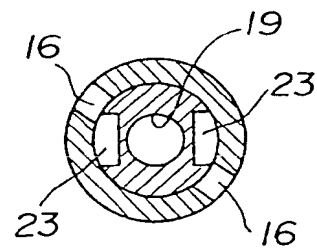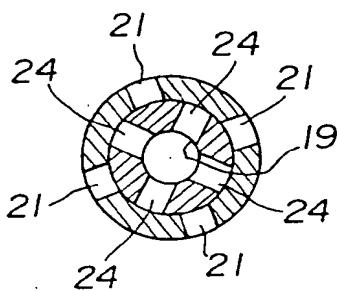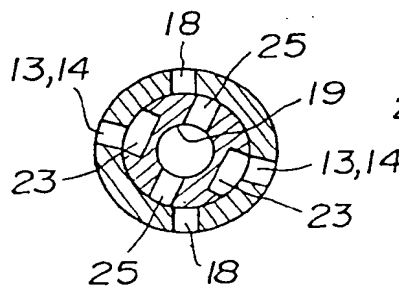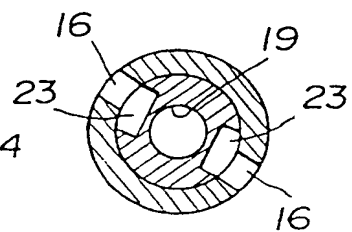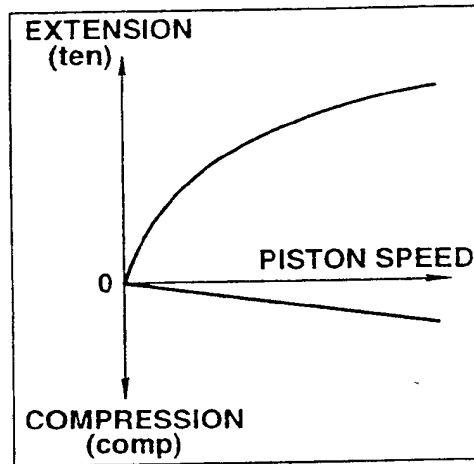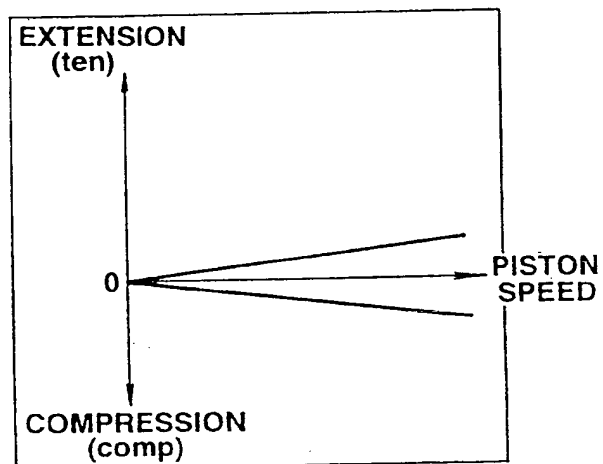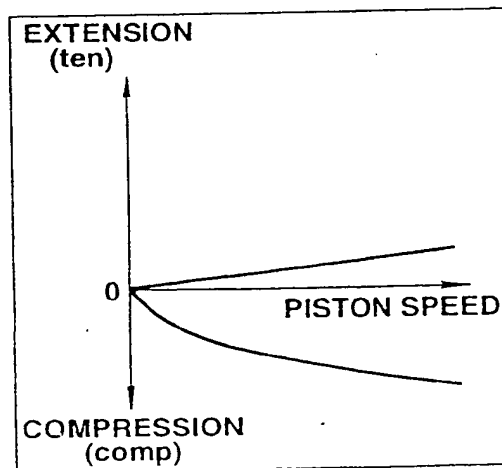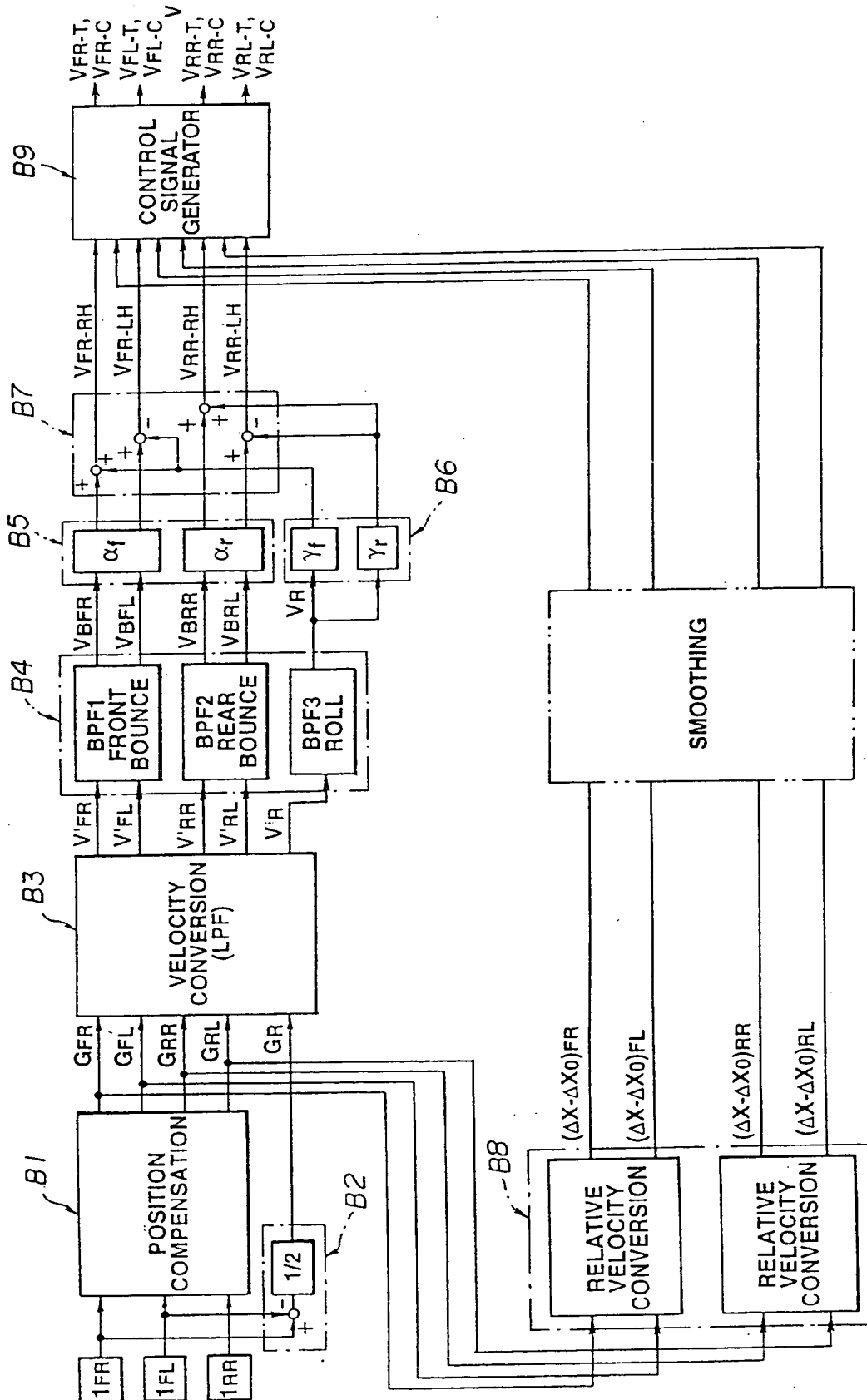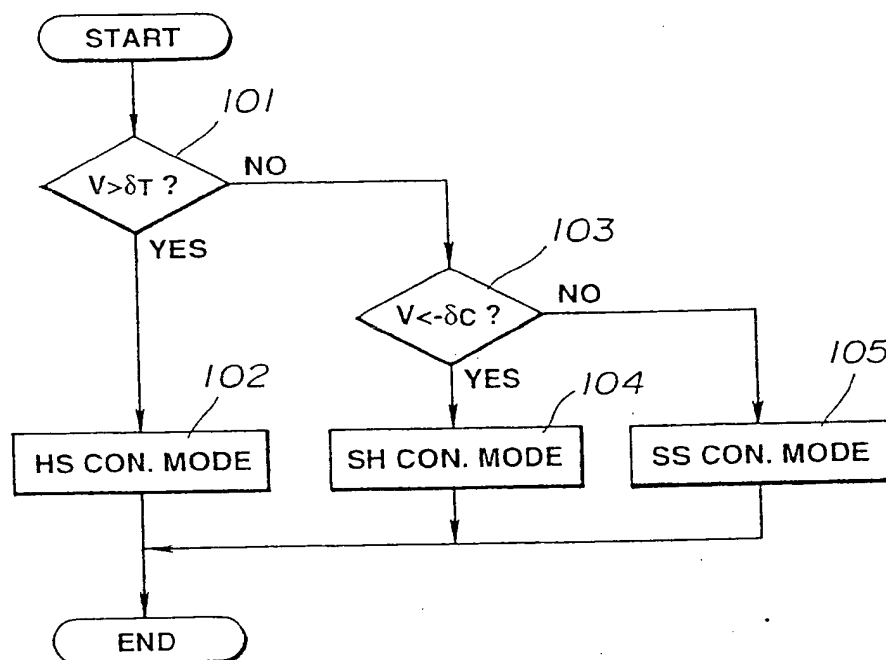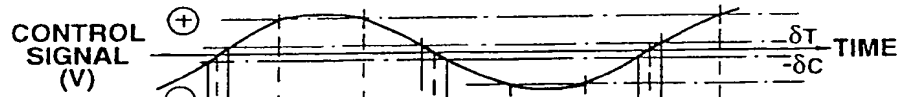
FIG.15A
FIG.15B
FIG.15C
FIG.15D
FIG.15E

FIG.16A SPRUNG MASS (VER.) VELOCITY (ΔX)

FIG.16B (ΔX-ΔX0) RELATIVE VELOCITY

FIG.16C (V) $\left[\frac{\Delta X}{\Delta X - \Delta X_0}\right]$ CONTROL SIGNAL

UPPER LIMIT

LOWER LIMIT

MAX POSITION

FIG.16D DAMPING FORCE CHARACTERISTIC POSITION (P)

MAX POSITION

FIG.16E DAMPING FORCE (C)

TIME

FIG.17A

FIG.17B

FIG.18

SPRUNG MASS → m1 ↑X1

SUSPENSION → { K1 C1

UNSPRUNG MASS → m2 ↑X2

ROAD WHEEL → { K2 C2 ↑X3

FIG.19A

(ΔX) → BPF [HPF (9Hz) → LPF (13Hz)] → $(\Delta X - \Delta X_0)$

FIG.19B

(ΔX) → HPF (10Hz) → $(\Delta X - \Delta X_0)$

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(54) **Apparatus and method for controlling damping force characteristic of vehicular suspension system**

(57)   In apparatus and method for controlling a damping force characteristic for a vehicular suspension system, a relative velocity between a sprung mass and an unsprung mass is calculated using a predetermined transfer function from a sprung mass vertical velocity derived on the basis of a sprung mass acceleration signal. The predetermined transfer function is, in a preferred embodiment, expressed as $G_{U(S)} = (QS^2 + RS)/(FS^2 + DS + E)$, wherein $Q = -m_1 \cdot c_2$, $R = -m_1$ $\cdot k_2$, $F = c_1 \cdot c_2$, $D = c_1 \cdot k_2$, $E = k_1 \cdot k_2$, S denotes a Laplace operator, c_1 denotes an attenuation constant of each of the shock absorbers, c_2 denotes an attenuation constant of the corresponding one of road wheels, k_1 denotes a spring constant of the shock absorber, k_2 denotes a spring constant of the corresponding one of the road wheels, m_1 denotes the sprung mass, and m_2 denotes the unsprung mass.

FIG.2

European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 95 11 5432

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
A	GB 2 263 959 A (TOKICO LIMITED) * page 4, last paragraph - page 5; claim 1 *	1,7	B60G17/015
A	EP 0 403 803 A (AUGUST BILSTEIN GMBH & CO) * column 3, line 7 - column 3, line 32 *	1,7	

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Attorney Docket No.   203-036

Application Serial No.   10/829,149

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
B60G

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
BERLIN	24 April 1997	Deprun, M

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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